UCRL-JC-125396
PREPRINT

Stability Design Considerations for Mirror Support Systems in ICF Lasers

G. L. Tietbohl
S. C. Sommer

This paper was prepared for submittal to the
2nd Annual International Conference on Solid-State Lasers
for Application to Inertial Confinement Fusion (ICF)
Paris, France
October 22-25, 1996

October 25, 1996

Lawrence Livermore National Laboratory

Stability Design Considerations for Mirror Support Systems in ICF Lasers

Gregory L. Tietbohl
Stanley C. Sommer

Lawrence Livermore National Laboratory
Livermore, California, 94551 USA

ABSTRACT

Some of the major components of laser systems used for Inertial Confinement Fusion (ICF) are the large aperture mirrors which direct the path of the laser. These mirrors are typically supported by systems which consist of mirror mounts, mirror enclosures, superstructures, and foundations. Stability design considerations for the support systems of large aperture mirrors have been developed based on the experience of designing and evaluating similar systems at the Lawrence Livermore National Laboratory (LLNL). Examples of the systems developed at LLNL include Nova, the Petawatt laser, Beamlet, and the National Ignition Facility (NIF). The structural design of support systems of large aperture mirrors has typically been controlled by stability considerations in order for the large laser system to meet its performance requirements for alignment and positioning. This paper will discuss the influence of stability considerations and will provide guidance on the structural design and evaluation of mirror support systems in ICF lasers so that this information can be used on similar systems.

Keywords: mirror, stability, structural support, ambient vibration, damping, laser, ICF

2. INTRODUCTION

ICF is the process of creating fusion in the laboratory using short-pulse, high-energy lasers that are focused onto targets in which the light energy implodes a small glass sphere containing isotopes of hydrogen. If the hydrogen atoms in the target are compressed for a sufficiently long period of time at the required temperature, helium atoms are formed and significant amounts of energy can be released. While fusion occurs naturally in stars, such as our sun, and is created in thermonuclear explosions, it has not yet been harnessed for use as a viable power source. To conduct high energy-density physics experiments for evaluating the use of fusion as a viable power source and for nuclear stockpile stewardship activities, ICF research has been ongoing for the past 40 years using increasingly larger laser systems.[1] These systems employ physically large optical components with clear aperture diameters greater than 400 millimeters and are contained in facilities which are hundreds of meters in length.

For the discussion in this paper, a mirror system consists of a mirror, mirror mount, mirror mount enclosure, superstructure, and foundation (as shown in Figure 1). A specific mirror system may contain more than one mirror on the superstructure. In large laser systems for ICF research, the beam size in a mirror support system is physically large (up to 1-meter diameter or rectangular edge length of 0.5 meters). An example of a mirror which directs a relatively large beam size is a 109-centimeter diameter turning mirror in the Nova facility[2] at LLNL. These mirrors are supported with customized mounts that are placed within relatively large mirror support systems. For the laser system itself, it is also physically large with components located at least 3 meters above a foundation and a beam path length among the laser components of hundreds of meters. One of the largest ICF laser systems which uses large aperture mirrors is NIF[3] which is currently being designed and evaluated at LLNL and is shown in Figure 2.

This paper summarizes some of the experience of designing and evaluating large laser systems for ICF research at LLNL. This experience includes information from operating systems such as the Nova laser system, the Petawatt laser, and Beamlet as well as the initial design effort (Title I) for NIF. There are many considerations for the structural design and evaluation of large laser systems including types of structural input, material types, and structural configurations. Based on the experience at LLNL, the design of the ICF laser systems has typically been controlled by stability considerations in order for the large laser system to meet its performance requirements for alignment and positioning. While the discussion in this paper will focus on the influence of stability considerations, general guidance about other structural considerations will also be provided. Smaller optical systems are not discussed in this paper because they are typically mounted on isolated optical tables and typically do not require superstructures for their structural support.

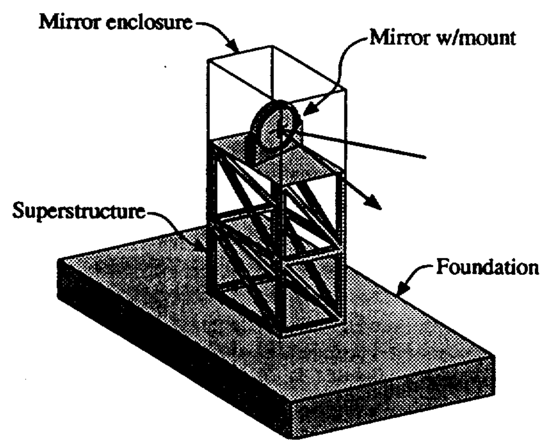


Figure 1. A mirror system consists of a mirror, mirror mount, mirror mount enclosure, superstructure, and foundation.

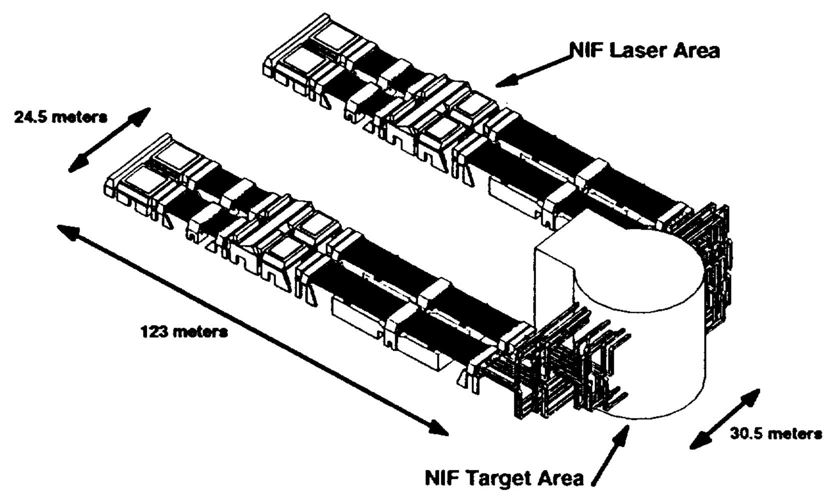


Figure 2. In order to achieve the performance requirements for the planned ICF research, the design of the National Ignition Facility results in a physically large system.

3. MIRROR STABILITY CONSIDERATIONS

3.1 Optical Alignment

ICF experiments with large laser systems use a short pulse length (one to twenty nanoseconds) which requires the mirrors in the system to be aligned properly and to remain stable with that alignment in order to position the beams on target as desired. The system must be aligned and then remain in that position for a period of time, such as two hours. It is also important to accurately align and center the beams in the laser system in order to minimize losses due to beam clipping, to maximize the laser performance at the target, to avoid diffraction patterns which result from the beam interacting with hard apertures, and to prevent laser-induced damage to hardware on the beam perimeter. The two major portions of alignment errors are in the near-field beam position and the beam position on target.[4]

For the near-field beam position, the major contributors are the accuracy of the centering steps in the alignment process and the incident angle of the amplifier slabs in the laser system. The requirements for the near-field beam position are typically contained as part of the clear aperture budget which considers many optical parameters in order to maximize the percentage of the laser optical surfaces including mirrors which are being used for the laser. After considering optical parameters, such as vignetting and Snell's walk, values in the clear aperture budget set the range of optical actuators and limit the displacement of optical components if they are subjected to static or long-term loads, such as dead weight or vacuum. For the design and evaluation of NIF, "the optical component placement allowance (X and Y) is set to ± 1 mm for pinholes, lenses, and light sources and ± 3 mm for other optical components."[5]

The major contributors to the beam position on target are the accuracy of the alignment process and the stability of the laser system before and during a shot. While the clear aperture budget provides the long-term positional requirements for the optical components such as mirrors, the beam position on target specifies how much that position is permitted to change before a shot and during the alignment process. Since mirrors are a critical optical component in the alignment process, they are a major contributor to the beam position on target requirement. In ICF experiments, the location of each focused beam on target must occur within a desired positioning error tolerance on the order of 50 microns. The beam position on target is an error summation of all the contributors which displace the beams from their appropriate locations at the target. These contributors include the steps in the alignment process which influence the accuracy of initial alignment, such as initial pointing and precorrection for beam steering, and the stability of the alignment in the laser system which is influenced by the structural drift of the optical components, such as the mirrors and spatial filter lenses. While translations of lenses modified by the focal length of the lenses relate to translations of the beams on target, rotations of mirrors multiplied by the focal length of the target result in translations of the beams on target. The contributions (Δx_{Target}) to the summation of the beam position on target are[4]:

$$\Delta x_{\text{Target due to Lens Motion}} = n * \Delta x_{Lens} * (f_{Target} / f_{Lens})$$

$$\Delta x_{\text{Target due to Mirror Motion}} = n * (2 * \Delta\theta_{Mirror}) * f_{Target}$$

where:

n - number of times beam passes through optic
Δx_{Lens} - displacement of lens surface
f_{Target} - focal length of target
f_{Lens} - focal length of lens
$\Delta\theta_{Mirror}$ - rotation of mirror surface

The alignment system will properly function only if the alignment accuracy and drift are carefully controlled. Development of the clear aperture budget discussed above is based on the centering drift being limited to a very small percentage of the optical component placement so that most of the clear aperture budget is available for optical considerations. If the drift is too large, the alignment system will be outside its compensation range. For NIF, about 80% of the error summation for the beam position on target results from considerations of structural drift while the remaining 20% accounts for alignment accuracy and contingency.[4]

3.2 Structural Drift

Using the Δx_{Target} terms discussed above, a flow-down for the structural drift portion of the beam position on target is determined by considering all the optical components that affect beam position on target. In the design and evaluation for NIF, the drift flow-down produces requirements that lens displacements be less than about 7 microns while mirror rotations be less than about 0.7 microradians. For multi-pass optical components, the drift limits are slightly less than for single pass components. The major contributors to the drift flow-down are the rotations of mirrors that reflect the beam twice (45% of the error summation) and the rotations of mirrors that reflect the beam once (about 30% of the error summation).[4]

The structural drift of the mirrors in a laser system is caused by external excitations including ambient vibration input, acoustical input, and thermal gradients. Similar to the beam position on target, the RMS deviation or drift of mirror rotations can be represented as an error summation of all the contributors as shown below:

$$\Delta\theta^2_{Total} = \Delta\theta^2_{Structural} + \Delta\theta^2_{Thermal} + \Delta\theta^2_{Contingency}$$

$$\Delta\theta^2_{Structural} = \Delta\theta^2_{\text{Random Vibration}} + \Delta\theta^2_{\text{Foundation Flexibility}} + \Delta\theta^2_{Mount}$$

where:

$\Delta\theta_{Structural}$ - rotations of mirrors contained in structural support systems
$\Delta\theta_{\text{Random Vibration}}$ - rotations of mirrors in a fixed-base superstructure that are caused by structural vibrations resulting from ambient motion, acoustical input, and vibratory input resulting from mechanical equipment

$\Delta\theta_{Foundation\ Flexibility}$ - rotations of mirrors in a superstructure caused by soil-structure interaction between the superstructure, foundation, and supporting soil

$\Delta\theta_{Mount}$ - rotations of mirrors caused by flexibility of optic supports or mounts between the superstructures and the optical components

$\Delta\theta_{Thermal}$ - rotations caused by thermal gradients in the superstructures and their associated components due to spatial and time variations in the air temperatures

$\Delta\theta_{Contingency}$ - contingency for design and evaluation unknowns

For most large laser systems, design requirements specify a thermally stabilized environment with relatively tight controls for the heating, ventilation, and air conditioning (HVAC) system. Since thermal gradients are difficult to predict and can constantly change, it is desired to have tight controls on the performance of the HVAC system or other heat-removal systems in order to minimize the influence of $\Delta\theta_{Thermal}$. With a temperature stabilized room, the LLNL experience has been that ambient vibration is the dominant contributor to $\Delta\theta_{Total}$ and the influence of the thermal gradients can be minimized.

3.3 Random Vibration Excitation

Random vibration excitation results from several sources including ambient vibration input, acoustical input, and vibrations from mechanical equipment. The excitation levels are dependent on the location of the laser system, the location of the vibration sources, and the design of the facility containing the laser system. In addition, large laser systems for ICF research have long beam paths so the effects of random vibration can be amplified through the length of the laser. Since random vibration excitation is a major contributor to $\Delta\theta_{Total}$, there have been considerable efforts to characterize and potentially modify the sources of the excitation. An important characteristic of random vibration excitation is that it is broad band (spread over a large frequency spectrum) with peaks at reasonances of the excitation sources. In addition, random vibration excitation is a very low level of input as compared with design levels of input from seismic motion or from static loads such as vacuum or weight. Materials typically respond elastically to random vibration excitation, which allows linear analysis to be used.

Ambient vibration input represents energy from the surrounding environment that is being transmitted through soil and is caused by many factors including road traffic, vibrations of nearby and distant mechanical equipment, microseisms at low frequency, and people traffic. This ambient input is then modified by the soil conditions and the foundation underneath the laser system. In general, the soil and foundation system deamplify the ambient input at low frequency and amplify the input at frequencies near the flexural frequencies of the foundation system. The amount of deamplification or amplification and the frequency ranges over which they occur depend on the relative stiffness of the foundation system and supporting soil. For some laser systems, its foundation is designed to isolate it from the soil so that ambient vibration input is deamplified over specific frequency ranges.

Random vibration excitation is also due to vibrations from mechanical equipment that is part of the laser system. To minimize the influence of vibration input and reasonances of most mechanical equipment, such as vacuum roughing pumps and air compressors, the equipment is usually isolated and located remotely. Some equipment, such as cryo-pumps, can be a source of low frequency excitation because they have to be mounted on or near the laser system. Another source of random vibration excitation is acoustical input mainly from air flow in the HVAC system and resulting flow-induced vibration. In many laser systems, acoustical input is deamplified by lowering the air flow speeds and by minimizing the number of large panels which can be excited by acoustics. For the design and evaluation of large laser systems at LLNL, efforts are made to minimize mechanical equipment noise sources and acoustical input such that the ambient vibration from environment dominates the levels of random vibration excitation.

Using sensitive accelerometers, random vibration input can be measured and is typically represented as a power spectral density (PSD) function since there are many possible sources of input contained in a large frequency range. A comparison of random vibration measurements from the top of foundation systems of three laser facilities at LLNL is shown in Figure 3. The Nova 2-Beam floor has the highest levels of excitation and the Nova 10-Beam floor has the lowest. The three facilities are physically close to one another (within 100 meters) at LLNL and are not adjacent to relatively noisy ambient vibration sources such as highways or factories, so the differences in the PSD data shown in Figure 3 is a result of the influence of the design of the facilities. The Nova 10-Beam room is a relatively stiff structure that is essentially a large concrete box (about 30-meters long, 15-meters wide, and 25-meters tall) with a 1.4-meter thick floor sitting on grade and 1.8-meter thick walls. In contrast, the Beamlet laser bay has a 0.9-meter thick, isolated slab supported by concrete columns above a basement. Finally, the Nova 2-Beam floor is a relatively flexible structure with a 0.3-meter thick, lightweight concrete floor supported by steel I-beam columns above a basement. Figure 3 also shows the PSD spectrum which is being used in the Title I design and evaluation of NIF to represent the random vibration at the top of the NIF foundation systems.[6]

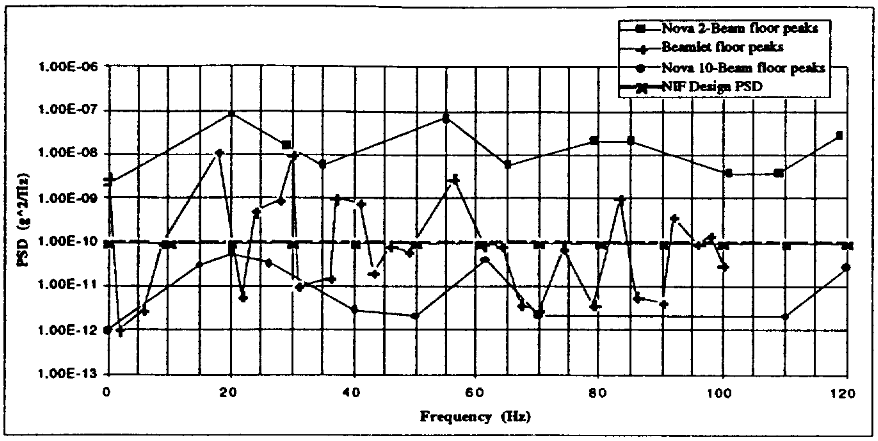


Figure 3. Power spectral density plots for random vibration in the vertical direction measured at the top of the foundation slab in different facilities. The data shown for NIF is being used in the Title I design and evaluation of the superstructures supported by the NIF foundation systems.

4. STABILITY OF SUPERSTRUCTURES FOR LARGE APERTURE ICF MIRRORS

4.1 Design Considerations for Stability

In large laser systems for ICF research, the mirrors are supported by superstructures which are relatively tall and open structures due to concerns with cleanliness control and access requirements. The mirrors in the NIF design are serviced from below which requires them to be at least 3.5 meters above the base of the superstructures. In addition, the majority of the mass supported by the superstructures is from the mirror components which are relatively heavy and are located near the top of the superstructure. As will be discussed in this section, it is desirable to have relatively stiff superstructures in order to satisfy the stability requirements for the mirrors and to minimize the amplification of input excitation at the support locations of the optics. The operational requirements for supporting large masses with tall and open structures result in challenging structural designs because tall and open structures are relatively flexible.

Previous large laser systems at LLNL were designed and evaluated to be relatively stiff by using stiff support members and materials and by incorporating as much bracing and shear panels as possible. Limitations imposed by cost and operational considerations typically resulted in superstructures with fundamental fixed-base frequencies in the 10 to 20 hertz range. In comparison, buildings typically have fundamental frequencies of about 2 to 5 hertz. The fundamental frequency is an important structural parameter because the dynamic response of the superstructure and the mirrors supported by the superstructure is dominated by this frequency. An example of a relatively stiff superstructure that uses stiff support members with significant bracing is shown in Figure 4. This superstructure supports the mirror mounts in the Petawatt vacuum compressor[7] and is a steel frame design with a fundamental frequency of about 15 hertz.

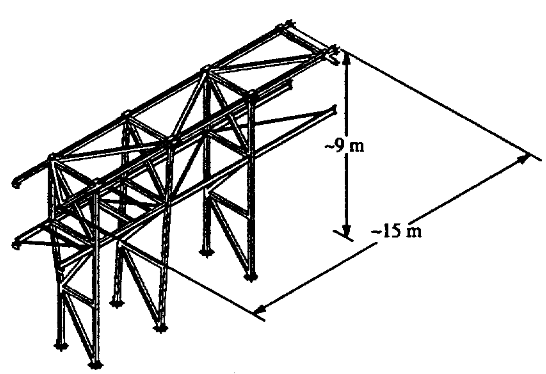


Figure 4. One example of a mirror superstructure, which was designed for the Petawatt laser at LLNL, is constructed of square steel tubing and I-beams. The superstructure connects to a concrete building at the base and at both ends.

The stability design of superstructures should have a balance of stiffness and damping. Stiffness is typically achieved with extensive bracing or large shear walls in structural frame systems. Damping, on the other hand, depends on the energy dissipation properties of the materials and connection details in the frame. For stiff frame systems, superstructures are often constructed of steel because it is inexpensive, available in pre-fabricated sections, and is fairly easy to construct and modify as required. While steel members are effective for stiff frame designs, steel has low modal damping at low levels of excitation such as ambient vibration. Recent designs for NIF superstructures incorporate hybrid structures of concrete and steel in order to increase the modal damping with concrete. As compared with steel, concrete is typically more expensive to achieve the same stiffness and is not fairly easy to modify once constructed. Other concepts are being evaluated that incorporate visco-elastic materials to increase system damping to even higher levels and these concepts are further discussed in Section 4.3.

While this paper focuses on the stability design of superstructures, there are other structural considerations which may impact the design. The effects of seismic input in areas of earthquake activity can be quite different than the effects of ambient vibration input since the level of excitation considered for seismic design is considerably stronger. Other structural loads such as vacuum loads and weight should also be considered as appropriate.

4.2 Evaluations of Designs

The design of superstructures supporting mirrors in large ICF laser systems has been controlled by the stability criteria for the mirrors. As discussed in Section 3, structural stability is typically the major contributor to the target error summation on beam position and ambient vibration is typically the significant portion of the external excitations. In order to evaluate the stability performance of a superstructure design, finite element analyses are typically performed. With finite element analyses, appropriate details of the superstructure, mirrors, and foundation should be included in order to evaluate the interactions between the different structural systems. Other techniques for the structural evaluations, such as analytical solutions and handbook approaches, should also be used for relatively straightforward analyses and to verify the results from the more detailed finite element analyses. A finite element model of a superstructure should include all the members which are part of the structural load path, the significant masses supported by the superstructure, appropriate boundary conditions, and interface conditions between different structural systems. Figure 5 shows a detailed finite element model of the NIF switchyard structure which supports many turning mirrors.

When subjected to ambient vibration input, the drift or rotation of a superstructure and its associated components ($\Delta\theta_{Structural}$) is a function of the mass and stiffness of the superstructure and components, of the system damping, and of the level of input. The relationship of the superstructure mass and stiffness is provided by the modes of vibration, or the structural frequencies. For most of the stiff and massive superstructures designed at LLNL, the dynamic response of the superstructures is dominated by the fundamental frequency, or the first structural mode of vibration, with almost 50% mass participation in this mode. This fundamental frequency of the superstructure also dominates the dynamic response of the entire mirror support system which includes the superstructure, its foundation, and mirror mounts.

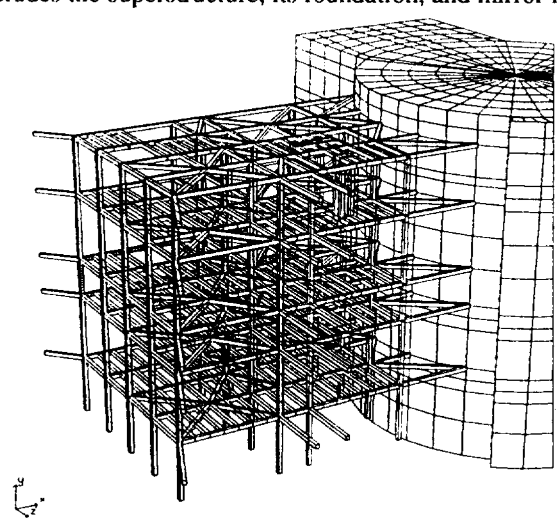


Figure 5. Finite element model of the NIF switchyard superstructure which consists of steel columns and beams for stiffness and is connected to concrete walls at the corners of the walls for stiffness and damping. The switchyard supports many turning mirrors throughout its height.

The overall frequency response of a mirror support system can be represented by a series of n springs:

$$K_{eq} = 1 / [(1 / k_1) + (1 / k_2) + ... + (1 / k_n)]$$

In the above relationship, the most flexible spring, or the spring with the lowest value of k, will dominate the response. For mirror support systems, the superstructure typically has the lowest stiffness especially if foundation flexibility is considered as discussed below. The mirrors, mirror mounts, and mirror enclosures typically have stiffness values which are five to ten times larger than the fundamental frequency of the superstructure. With a finite element model, the fundamental frequency and other structural frequencies of the superstructure and its associated hardware are computed with an eigenvalue extraction. For adequate dynamic representation of the modeled structures, the mass participation in the eigenvalue extraction should exceed about 80%. Some superstructure designs are highly stiff in the vertical direction, so it may be very difficult to obtain vertical mass participation above about 60%.

The response ($S_y(\omega)$) of a structural system to random excitation input ($S_x(\omega)$) is computed from the equation[8]:

$$S_y(\omega) = |H(\omega)|^2 * S_x(\omega)$$

where:

$H(\omega)$ - the transfer function between the location of the input motion and the location at which the response is being computed

Figure 6 shows an example of a transfer function which describes the dynamic relationship between a point on a mirror supported in the NIF switchyard structure shown in Figure 5 and the base of the switchyard. There are several peaks in Figure 6 below ~15 hertz that are representative of the structural frequencies of the switchyard. As expected, the largest peak is associated with the first fixed-base structural mode of the switchyard which occurs at 7.3 hertz. The NIF switchyard has a fundamental frequency below 10 hertz due to its size, the amount of steel decking that the superstructure is supporting, and access constraints. By 30 hertz, the magnitude of the transfer function has decreased by several orders of magnitude.

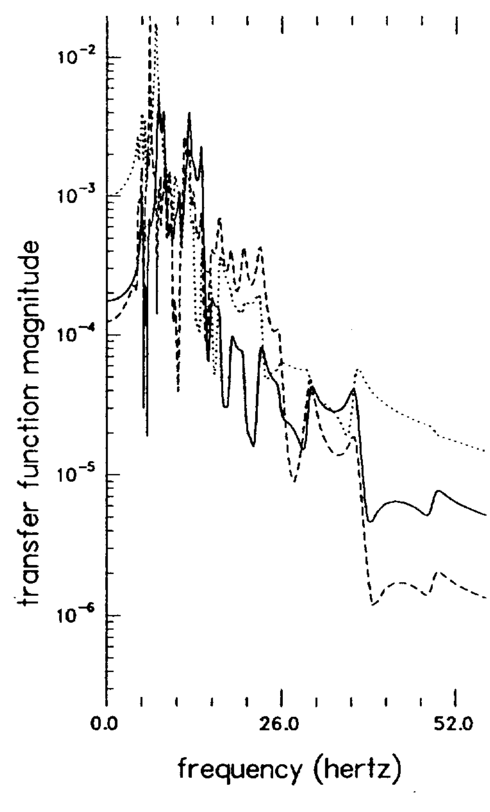


Figure 6. Transfer function for a turning mirror in the NIF switchyard between the base of the switchyard and the location of the mirror

To compute the response of the superstructure and its associated hardware to random vibration input, the results of the eigenvalue extraction are used. For vibration input at the base of the superstructure, the eigenvalues provide the information needed for computing the transfer functions ($H(\omega)$). Since the ambient vibration input ($S_x(\omega)$) is represented as a PSD, the output ($S_y(\omega)$) is also a PSD. The mean square response ($E[y^2]$) is defined as the integral of the response ($S_y(\omega)$) over all the frequencies of the structural system, or the area underneath the response curve. The mean square response is a statistical quantity and the square root of the mean square response is the displacement or rotation of the mirror ($\Delta\theta_{Random\ Vibration}$). Each direction of ambient vibration input produces three orthogonal directions of PSD output. These three directions of output are then statistically combined with the square-root-sum-of-the-squares technique, which assumes there is not phasing between the modes, or with other combination techniques which consider the modal phasing.

Typical models of the superstructures contain multiple degrees-of-freedom. To gather some insight about the response of the superstructure to ambient vibration input, the superstructure can be represented as a single-degree-of-freedom (SDOF) system with a structural mode which is the fundamental frequency of the superstructure. The mean square response of a SDOF system to ambient vibration input with a constant amplitude is[8]:

$$E[y^2] = (\pi / 2) * (S_o / \xi * \omega_n^3) = (W_o / 8) * [1 / (\xi * (2 * \pi * f_n)^3)]$$

where:

S_o - amplitude of ambient vibration input
ξ - modal damping
ω_n - frequency in units of radians/second
W_o - amplitude of ambient vibration input in units of acceleration^2 / hertz
f_n - frequency in units of hertz

Using the above equation for a superstructure with a fundamental fixed-base frequency of 15 hertz, modal damping of 0.5% (nominal value for steel), and ambient vibration input of 1×10^{-10} g^2/hertz, the mean square response is 0.29 μm^2 and the displacement (square root of mean square response) is 0.54 microns. The relationship of the fundamental frequency of the superstructure and its response to ambient vibration input is clearly seen in the above equation. As the fundamental frequency decreases, the mean square response increases with the cube of the frequency.

The above equation also shows the relationship of the modal damping of the superstructure and its response to ambient vibration input. As the modal damping increases, the response decreases with the square root of the damping. For the Title I design and evaluation of NIF structures subjected to ambient vibration levels of input, nominal modal damping values of 0.5% for steel and 2.0% for concrete are being used,[6] although the value for concrete may be reduced to 1.0% in future analyses. The benefit of using concrete in the superstructures is demonstrated by the 4x increase in the modal damping as compared to steel, which results in a 2x decrease in the displacement of the superstructure. The use of passive damping, which is discussed in Section 4.3, can greatly decrease the displacement in the superstructure as well. Modal damping values of 5% to 10% with passive damping can provide a significant reduction in the displacement of the superstructure. For hybrid superstructures of concrete and steel or other materials, composite modal damping should be used when computing the response to ambient vibration input. Composite modal damping allocates the appropriate level of damping depending on the stiffness and mass of the participating structural elements in a particular mode (e.g., accounts for the differences between steel and concrete).

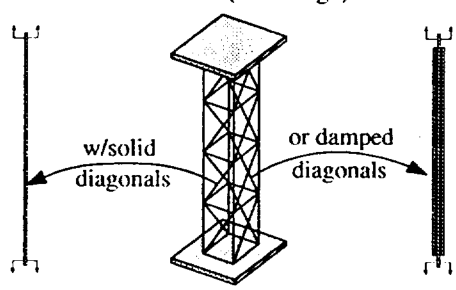
A significant consideration for the design and evaluation of superstructures for large ICF laser systems is the foundation systems supporting the superstructures and mirrors. As the laser systems become larger and the superstructures become more stiff and massive, the structural demands on the foundation systems increase dramatically. In general, it is desirable for the foundation system to be a slab on grade instead of an elevated structure. An elevated structure will have its own dynamic behavior which can result in magnification of the ambient vibration input. Another advantage of a slab on grade is that isolation can be used between the slab and the supporting soil in order to reduce the ambient vibration input. It is also desirable for the foundation system to be as stiff as economically feasible, and other foundation systems, such as pile foundations, have been considered for stiffening superstructures. Finally, the foundation system for the superstructures and optical components should be isolated from the foundation system of structures supporting the mechanical equipment.

Typically, the finite element analyses of the superstructures yield the fixed-base frequencies in which the base of the superstructures is assumed to have fixed translations and rotations. The effects of foundation flexibility resulting from the interaction of the superstructure with its foundation and supporting soil should also be included when evaluating the rotations of mirrors in the superstructures ($\Delta\theta_{Foundation\ Flexibility}$). As part of the NIF Title I design and evaluation activities, this interaction of the massive and stiff superstructures on the relatively flexible slab foundation and its supporting soil has been evaluated. With foundation flexibility, the fundamental frequency of the superstructure decreases about 30% and this decrease causes the displacements of the superstructure to increase by about a factor of 1.3 to 1.5.[9]

4.3 Passive Damping

Design of a passively damped system can significantly increase the modal damping of a steel structure by a factor of ten or more. Specific details on this concept are beyond the scope of this paper, but it involves the use of viscoelastic material in the structure to dissipate energy (essentially a damper in parallel with a spring). This concept can be used in bending or truss-type structures and affects a wide frequency range at low excitation levels. Consideration must be given to heating of the viscoelastic material from high excitation levels such as seismic loads and to the effects of fire on structural stability. If a superstructure cannot meet performance specifications, an increased value for the system damping (to 5% or 10%) can be used when performing the finite element frequency analysis. If the resulting drift of the superstructure and mirrors are acceptable, then experts with passive damping experience can be consulted to design a specific passive damping system (CSA Engineering[10], 3M Engineered Materials[11]).

One example of a passively damped truss tested at LLNL is shown in Figure 7. Tests were performed using a small truss structure (1.2 meters high) in which the solid metal diagonals were easily replaced with passively damped ones. The damped diagonals consisted of a continuous load carrying member with transfer elements applied to two sides and viscoelastic material in between. This diagonal design acts as a spring whose ends impart shear in the viscoelastic material as a result of motion. The transfer elements must be comparably stiff to the diagonal elements in order to resist the shear of the viscoelastic material. A significant increase (4x to 40x) in the modal damping of the structure was obtained at low levels of excitation (<0.01 g).



Modal Damping w/Baseline Diagonal	Vibrational Mode	Modal Damping w/Damped Diagonal
0.7%	First bending	2.6%
0.1%	First torsional	4.6%
0.3%	Second bending	3.3%

Figure 7. Passive damping of a small truss structure yielded a significant increase in modal damping at low excitation levels (<0.01g).

A tuned mass damper is another passive damping technique that is used to move a natural frequency of structure away from a particular frequency, such as one at which an excitation spike exists. This technique has not been employed with the laser systems at LLNL since the random vibration input spectrum is generally a broad-band noise source, but this is a viable damping technique and should be considered when appropriate.

5. MOUNT DESIGN FOR LARGE APERTURE ICF MIRRORS

As discussed earlier, mirror mounts should be designed with a fundamental frequency above 50 hertz (preferably in the 100 to 200 hertz range). It is generally easier to obtain a high fundamental frequency of vibration for a smaller system, such as a mirror mount, due to the larger ratio of stiffness to mass. The maximum fundamental frequency that can be obtained for a particular mirror mount is sometimes limited by space and mass constraints.[12] A gimbal-type mount (Figure 8) has two concentric rings that can be independently rotated to provide tip/tilt capability. There are many other types of mirror mounts that have different adjustment concepts.[13]

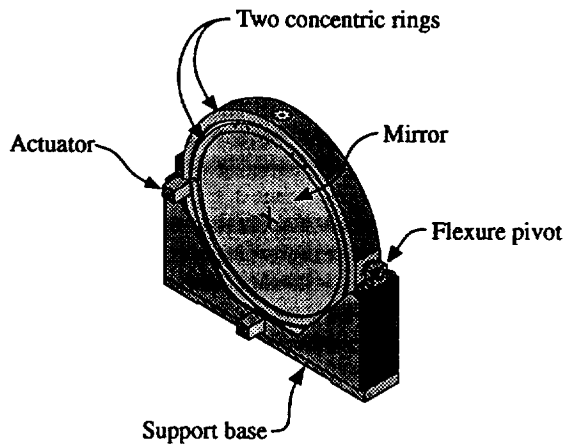


Figure 8. A mirror mount for a 94-centimeter diameter optic uses flexure pivots.

To provide a stable, distortion-free mount, it is preferable to support the mirror at three points so that no moments are imparted. The mirror surface must remain optically flat with the mirror supported in its mount. If the mirror is mounted with its surface near a vertical plane, finite element analysis can predict the out-of-plane distortion of the mirror surface that results from the weight of the mirror resting on its supports (this should be $<\lambda/50$). If mounted horizontally, gravity sag of the middle of the mirror can be significant (one or two waves) if perimeter support is used. An inboard wiffle tree design will reduce gravity sag, but this concept is not feasible if a transmitted beam is required for diagnostic purposes.

The adjustment system that provides the tip/tilt capability is critical for achieving a high frequency of vibration of the mirror assembly. It is preferable to utilize flexure connections with rigid clamps wherever possible. However, flexures limit angular motion to about $\pm 3°$. Bearings should be avoided, but high precision bearings (ABEC 7 or 9) should be used if necessary with an interference fit. Stiction of the bearings can often limit the tip/tilt resolution achievable. To accommodate the tip/tilt adjustment, a ring-mounted mirror (gimbal type) should be held at three points such that two points define a rotation axis and a third point (actuator) defines the plane. The actuator design must resist axial motion (no slop) which would otherwise lower the fundamental frequency of vibration.

The gimbal-mount type of support results in a "diving board" fundamental mode of vibration of the mirror, since the mirror edge opposite the actuator is not supported. The stiffness of the support ring can therefore be important. Mounts with three independent actuators behind the mirror can provide higher fundamental frequencies of vibration, but are more difficult to design. It is generally easier to design a stable mount for round mirrors than for square ones, due to the inherent bending flexibility and torsional weakness of rectangular frames.

A design of a mirror mount can be evaluated using finite element analysis to determine its natural frequencies of vibration, as discussed for the superstructures. Care must be given in creation of the model to properly account for connections, including support points and boundary conditions. A finite element model of the mirror mount assembly is created to determine the natural frequencies of vibration, the model is modified, and the process is repeated as necessary until the desired result is obtained. Calculation of rotations of the mirror due to random vibration input ($\Delta\theta_{Mount}$) is necessary if the mirror mount is relatively flexible (fundamental frequency less than 50 hertz) and to capture the kinematics of the connections between the mirror mount and the superstructure. If the mirror mount has a fundamental frequency of vibration that is significantly higher than the superstructure that supports it, the flexibility of the mirror mount will not influence the rotations of the mirrors. The kinematics of the connection between the mirror mount and the superstructure may influence the rotations of the mirrors and these connections should be appropriately considered in the finite element analyses of the mounts and the superstructures. If appropriate, the mirror mount should also be designed so that it is structurally secure in a seismic event so that the mirror will not be damaged and the mount will not become dislodged.

6. MIRROR ENCLOSURES FOR LARGE APERTURE ICF MIRRORS

Enclosures should be designed to surround mirror mounts to reduce air currents that can affect beam stability and to improve cleanliness conditions around the mirror which reduces the likelihood of damage and lengthens mirror life. As discussed earlier, the room air handling system will often have noise sources in the HVAC ducting that produce an acoustic signature in the room. The enclosure panels can act as diaphragms which are easily excited due to the large surface area and low inherent damping. Vibration of the panels can then be transmitted into the superstructure and on into the mirror mount. This acoustic coupling can increase the motion of the mirror resulting from other random vibration sources. If the panel has natural frequencies of vibration close to the superstructure frequencies, this motion will be amplified.

The acoustic energy that is coupled to the panels can be dissipated using passive damping techniques. Mirror mount enclosures should incorporate passively damped panels which are inexpensive and significantly reduce the acoustic coupling to the structure. Passively damped panels can be easily constructed using a layer of viscoelastic material between two layers of sheet metal (see Figure 9). The energy absorbing medium is placed at the location of highest shear stress which produces an acoustically "dead" panel. Use of this technique on the Beamlet laser at LLNL using a commercially available viscoelastic material[14] has helped reduce mirror motion and improved focal spot stability. In addition, enclosure panels are usually not considered structural elements in the superstructure design and analysis, but can be included if they are sufficiently thick and are well connected to the structure. If appropriately designed and constructed, these enclosure panels can provide considerable shear stiffness to the superstructure.

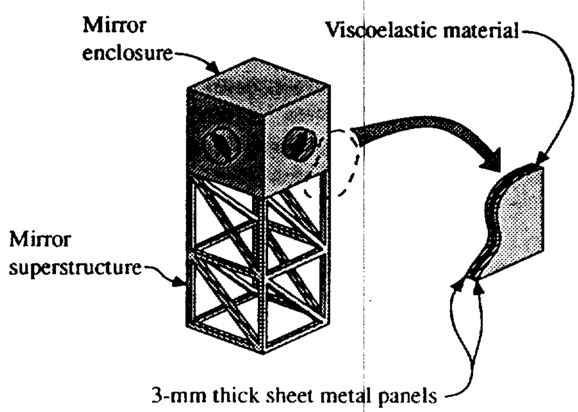


Figure 9. Highly damped panels can be easily added to a mirror mount enclosure as shown in this figure.

7. SUMMARY

Based on the experience of designing and evaluating mirror systems to remain stable in large ICF lasers at LLNL, the following is recommended for similar systems:

- Design the superstructure with a high fundamental frequency of vibration (>10 hertz).
- Design individual mirror mounts for a high fundamental frequency of vibration (50 to 200 hertz).
- Incorporate, if possible, a rigid and isolated foundation system.
- Consider the stiffness of the foundation system in stability analyses depending on the stiffness of the mirror superstructure relative to the foundation system stiffness. The foundation system includes both the foundation and its supporting soil, so soil-structure interaction analyses may be necessary.
- Add passive damping to the structure if necessary to increase energy dissipation and to reduce motions.
- Use passively damped panels for mirror enclosures.
- Maintain temperature stability of the room.

8. ACKNOWLEDGMENTS

This work was performed at Lawrence Livermore National Laboratory under the auspices of the United States Department of Energy, Contract Number W-7405-ENG-48.

9. REFERENCES

1. R.R. Johnson, "Laser Fusion Efforts in the United States", *Optics & Photonics News*, **6**(3), 16-23, March 1995.
2. E. M. Campbell, T. J. Hunt, E. S. Bliss, D. R. Speck, R. P. Drake, "Nova Experimental Facility", *Review of Scientific Instruments*, **57**, 2101-2108, 1986.
3. National Ignition Facility Title I Review, Lawrence Livermore National Laboratory, Livermore, CA, 1996.
4. National Ignition Facility Conceptual Design Report, NIF-LLNL-94-113, Lawrence Livermore National Laboratory, Livermore, CA, May 1994.
5. R. E. English, "Clear aperture budget for main laser system (LM1-SF4)", NIF-0000475 REV 1, Lawrence Livermore National Laboratory, Livermore, CA, August 26, 1996.
6. S. C. Sommer, "Ambient Ground Vibrations and Structural Damping Values", NIF-LLNL-96-141, Lawrence Livermore National Laboratory, Livermore, CA, February 26, 1996.
7. "Petawatt Laser Final Report", UCRL-ID-124933, Lawrence Livermore National Laboratory, Livermore, CA, October 1996 (not yet released).
8. S. H. Crandall and W. D. Mark, Random Vibration in Mechanical Systems, Academic Press, New York, NY, 1963.
9. M. Tabatabaie, "Analyses of Soil Structure Interaction due to Ambient Vibration - Proposed National Ignition Facility, Livermore, California", Kleinfelder, Inc., Pleasanton, CA, August 31, 1996.
10. CSA Engineering, Inc, 2850 W. Bayshore Road, Palo Alto, CA, 94303, USA, phone: (415) 494-7351.
11. 3M Engineered Materials, 3M Center, Building 220-8E-04, P.O. Box 33220, St. Paul, MN, 55144, USA, phone: (612) 736-4664.
12. R. M. Hamilton, R. and G. L. Tietbohl, "Design of a ϕ94 cm Mirror Mount For the Petawatt Project on Nova", UCRL-ID-122342, Lawrence Livermore National Laboratory, Livermore, CA, October 1995.
13. L. C. Hale, "Guidelines for Design and Analysis of Kinematic Devices", NIF-LLNL-96-0000210, Lawrence Livermore National Laboratory, Livermore, CA, June 25, 1996.
14. Avery FasTape^{tm} 3099 UHA, Avery Dennison, Specialty Tape Division, 250 Chester Street, Painesville, OH, 44077, USA, phone: (216) 639-2600.